EFFECT OF TWO STRESS RAISERS ACTING TOGETHER AT A POINT

DONALD E. GUHSE

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Commander, United States Navy

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the requirements for the degree of

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ABSTRACT

Stress concentration factors are valuable tools for the machine designer when they can be accepted as reliable. When one stress concentration configuration, such as a hole, fillet, notch, machine finish, etc. exists there are numerous curves and equations available that give the theoretical extent of concentration to be expected. With two or more of these configurations present at the same point, the designer must decide whether or not to combine factors, and if so, in what manner, to obtain a good prediction.

Fatigue tests in torsion were conducted in order to search for an answer to the question of a reasonable design basis for repeatedly stressed points with two stress raisers.

The author wishes to acknowledge the valuable assistance and encouragement given him by Professor Virgil M. Faires, of the U. S. Naval Postgraduate School, during this project.

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1. Introduction

The problem of improved design of machine parts continually confronts the engineering profession. Untimely failure of parts is costly from the standpoint of cost of replacement, loss of production time, and, more seriously, injury or loss of life. On the other hand, use of excessive safety (or ignorance) factors is becoming less desirable due to weight limitations, limited availability of some materials, and the necessity of limiting material and production costs to meet competition.

Improper design is involved in the majority of premature failures. The situation is well stated as follows in a publication of the U. S. Navy Bureau of Aeronautics[1]:

> While minor improvements in fatigue life may be accomplished merely by changing material, few serious fatigue difficulties have been completely corrected in this way.

The publication also states:

> By studying stress concentration factors much can be learned about how to produce designs that are superior from the standpoint of resistance to repeated loads and how to evaluate approximately the influence of various geometric figures.

The general analytical solution of fatigue failures has yet to be developed, but much empirical and experimental data have been compiled over a period of years. The design engineer's basic tool, the "stress concentration factor", is available in the form of charts, curves, tables, and equations. A comprehensive and valuable collection of charts and relations useful in making strength calculations has been compiled by Mr. R. E. Peterson [1].* This book, as well as other literature on the

* Numbers enclosed in brackets [ ] refer to bibliography on page 36.

[1] R. L. Templin and E. C. Hartmann, Designing for Repeated Loads, NavAer SM-32, Jan. 30, 1952.

subject, deals only with stress-concentration configurations appearing singly, and does not treat situations where two stress concentrations appear together at a point.

The problem of combined stress raisers has generally been solved by: tests of actual or minaturized parts under maximum anticipated loads, combination of known stress concentration factors based on experience of individual designers or manufacturers, application of large safety factors, and other methods of questionable reliability. A view of the current situation is expressed by Dr. Horace J. Grover of Battelle Memorial Institute in the following quotation[2]:

> Composite structures usually involve uncertainties in detailed stress analysis plus inadequate information concerning effects of stress concentrations such as those around discrete fasteners in joints. For present design, a useful empirical procedure involves experimental stress analysis under static loading plus fatigue tests of specimens carefully designed to include local stress-concentration effects. Future analysis can be aided by investigations providing more understanding of effects on fatigue behavior of some of the types of stress concentrations that exist in composite structures. Items for which information seems particularly inadequate include: joints with rivets (and bolts or other discrete fasteners) under complex loading, fretting, etc. For composite structures, use of the concept of an effective stress-concentration factor requires extreme care.

In regard to the object of this project, a statement by Mr. E. C. Hartman, reporting on the effect of unintentional stress raisers on the fatigue strength of structural components, seems appropriate:[3]

[2] H. J. Grover, Allowance for Stress Concentration in Design to Prevent Fatigue, Proceedings of the International Conference on Fatigue of Metals, Institute of Mechanical Engineers, London, 1956, p 88.

[3] E. C. Hartman, Effect of Unintentional Stress Raisers on the Fatigue Strength of Structural Components, Proceedings of the International Conference on Fatigue of Metals, Institute of Mechanical Engineers, London, 1956, p 199.

> If a discontinuity happens to be located and oriented exactly so as to intensify a high stress concentration caused by some design feature, then its effect is likely to be a real factor in determining the fatigue strength of the part.

(He defines a discontinuity as a scratch, gouge, cut, nick, undercutting, or an unwanted hole, in addition to one of a metallurgical nature.)

This investigation deals with a condition where two discontinuities, a hole and a rough machined surface, appear together. Fatigue tests in torsion were conducted to obtain an indication of the relationship between the stress concentration factor for a hole versus the stress concentration factor for a hole and a rough machined surface combined. The effect of the combined stress raisers was also compared with the stress concentration factor for a rough machined surface.

The tests were conducted by the author in the Mechanical Engineering Laboratory of the U. S. Naval Postgraduate School, Monterey, California, under the supervision of Professor Virgil M. Faires during the period from December 1959 to May 1960.

2. Equipment

Specimens.

Selection of the type of material to be used in the tests was based on the decision to use a deep hardening steel that is heat treatable to a surface hardness of about 350 Brinell. All specimens were fabricated from four 15-foot bars of 1-1/2 inch round stock, AISI 4140 steel produced in an electric furnace, hot rolled, and normalized. The four bars were certified to be from the same heat and to contain the following percentages of alloying elements by weight: 0.39% carbon, 0.90% manganese, 0.009% phosphorus, 0.019% sulfur, 0.27% silicon, 0.90% chromium, and 0.20% molybdenum. In the normalized condition, the steel was certified to have the following physical properties: yield point, 102,500 psi; tensile strength, 125,000 psi; Brinell hardness, 269; ASTM grain size, 7-6; and Jominy, 5-56 10-48.

As each specimen length was cut from the bar stock, it was identified by letter (A,B,C, or D) to indicate which bar it came from and by number (1,2,3, - -, 21) to indicate its original location in the bar. These pieces were rough machined to within about 1/16-inch of the final dimensions and were then heat treated. The heat treatment was accomplished by the author, utilizing the facilities of the Metallurgical Laboratory of the U. S. Naval Postgraduate School. The desired heat treatment was ascertained from the characteristic curves for AISI 4140 steel shown in figure 1[4]. Four or five specimens were placed in an electric furnace set at 1550°F (± 5°) for 2-1/4 hours, quenched in agitated oil, tempered in an electric furnace at 1000°F (± 5°) for 1-1/4 hours, and finally air quenched. The specimens were then finish machined to the desired configuration. It was expected that this procedure would remove surfaces

[4] Bethlehem Steel Company, Modern Steels and their Properties, Handbook #268, 1949, p 114.

AISI - 4140

(Oil Quenched)

PROPERTIES CHART
(Single Heat Results)

			C.	Mn.	P. Max.	S. Max.	Si.	Ni.	Cr.	Mo.	Grain Size
.530" Rd. SIZE TREATED / .505" Rd. SIZE TESTED	Ac_1 1395°F. / Ac_3 1450°F.	Ar_3 1330°F. / Ar_1 1280°F.	.38/.43	.75/1.00	.04	.04	.20/.35	Nil	.80/1.10	.15/.25	
HEAT TESTED			.41	.85	.024	.031	.20	.12	1.01	.24	6-8

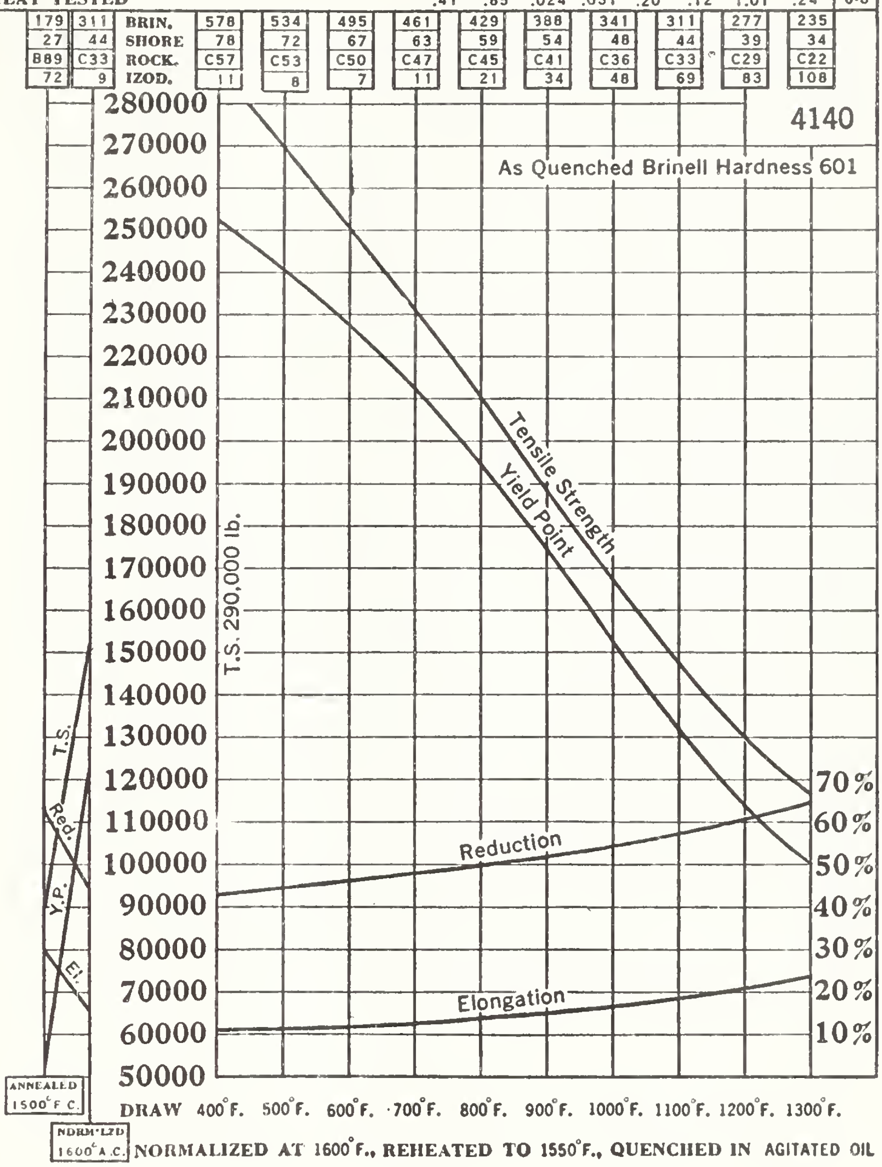


Fig. 1

decarburized during heat treatment. A 0.040-inch mill cut was made the full length of one shank on each specimen to provide a flat for hardness readings.

Six specimens were initially fabricated in an effort to determine the optimum set of configurations for prosecution of this project. They were all patterned after the standard "M" type specimen, figure 2, recommended for torsion testing on the Baldwin Universal Fatigue Testing Machine[5]. These specimens had rough and smooth finished surfaces in the following shapes: a) standard solid, b) standard with 1/16-inch transverse hole, and c) standard specimen modified to incorporate a 1/8-inch radius fillet.

The types of specimens tested in this project were chosen after preliminary tests indicated sufficient variation in fatigue life to give S-N curves significantly different for each configuration. These configurations were also chosen because they could be readily reproduced in the U. S. Naval Postgraduate School Machine Shop. The four basic configurations, shown in figure 3, were: (1) standard specimen with a smooth (4 to 8 microinch) finish, (2) standard specimen with a rough machined (250 microinch) finish, (3) standard smooth finish with a 1/16-inch transverse hole through the narrow section, and (4) standard rough machined finish with a 1/16-inch transverse hole through the narrow section.

[5] Baldwin Locomotive Works, Instructions for Operation of Universal Fatigue Testing Machine Model SF-1-U, Sept. 26, 1946.

TYPE "M" TORSION SPECIMEN

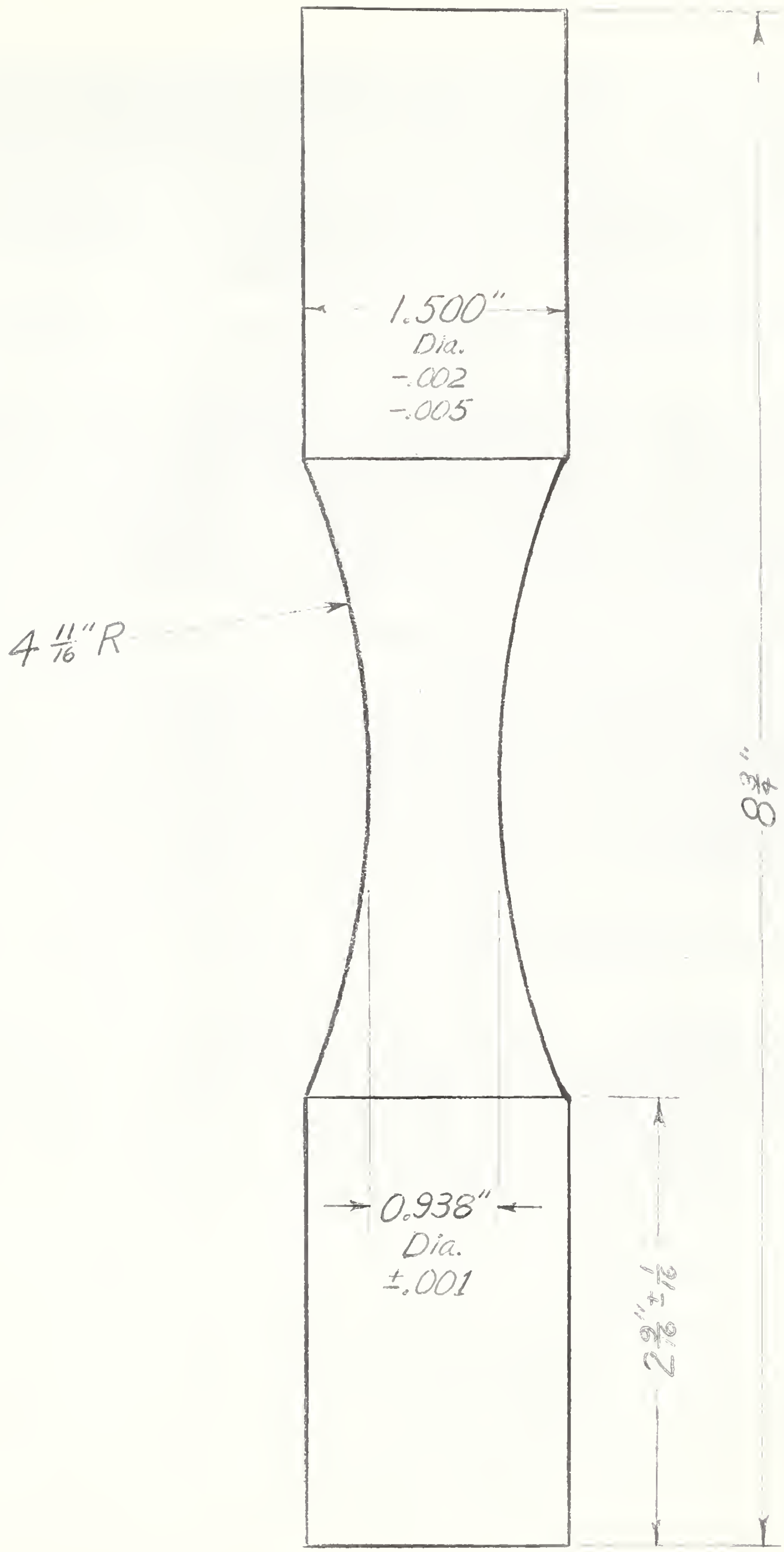
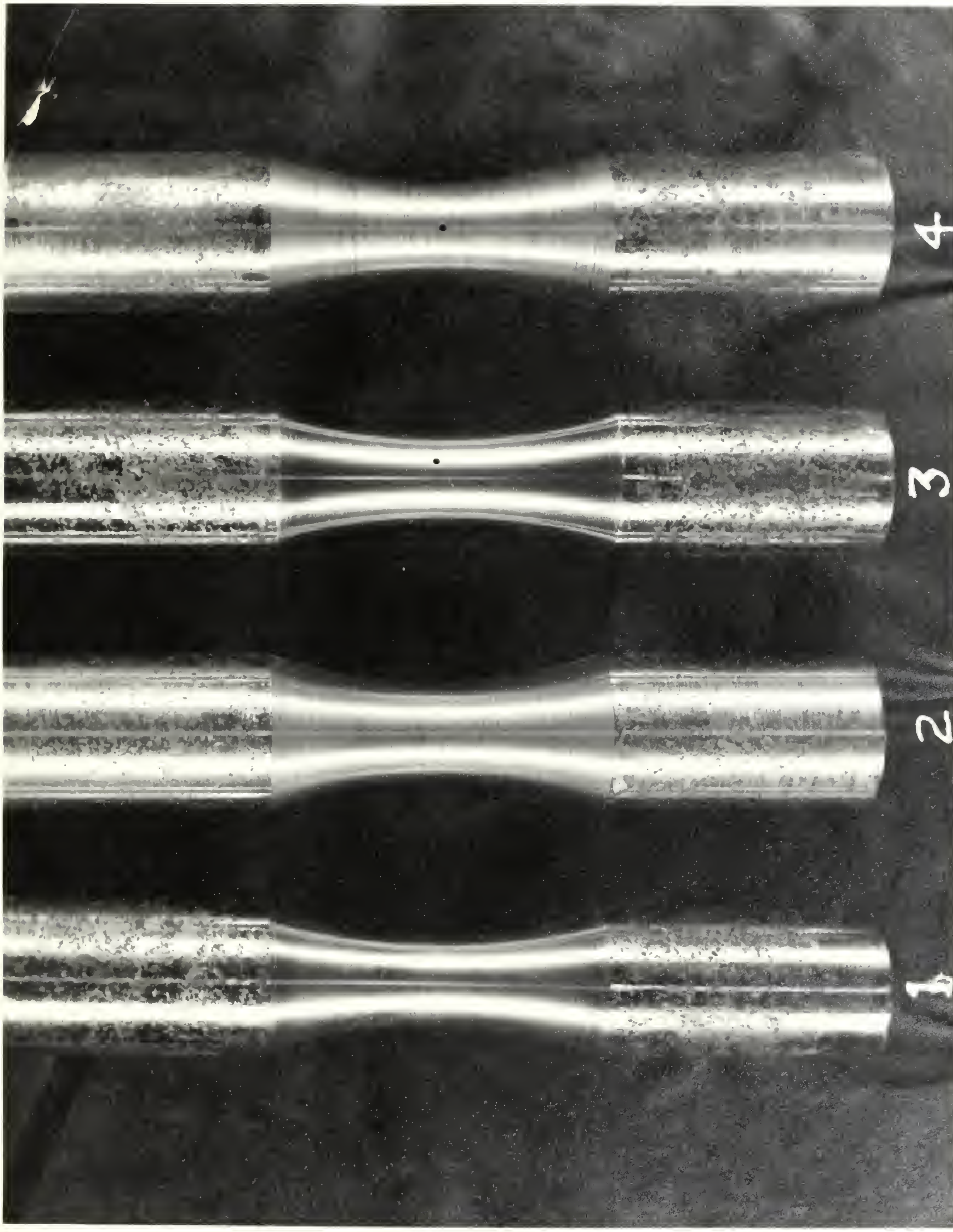


Figure 2


1
2
3
4

The finishes indicated above were determined by visual and fingernail comparison with standard finishes.

In addition to these basic configurations, a group of four standard rough machined specimens, figure 4, with a 1/4-inch transverse hole in the narrow section were made, and a group of seven smooth finish specimens of smaller dimensions, figures 5 and 6, were made from specimens that had been tested for 5 million cycles without failure.

Testing Machine.

The Baldwin Universal Fatigue Testing Machine (Sonntag), model SF-1-U, figure 7, installed in the Mechanical Engineering Laboratory of the U. S. Naval Postgraduate School, was used for all fatigue tests. The machine is designed to apply a vertical vibratory force to any specimen or structure attached between the heavy stationary frame (E) and the reciprocating platen (F). The alternating force is produced by an unbalanced rotating mass, which is supported between two bearings in a cage-like vertical frame, the top of which forms the reciprocating platen. The rotating mass is driven by a synchronous motor so that its speed is maintained constant at 1800 revolutions per minute. The load may be adjusted by moving the eccentric weight along its displacement scale, which, according to the manufacturer, is accurate within 2% throughout the range of the scale.

The fixture shown in figure 8 is the standard fixture provided by the manufacturer for torsional fatigue testing of cylindrical specimens. The alternating force produced by the reciprocating platen (F) is converted into an alternating torque in the specimen by means of a crank (R), which is pivoted at one end and oscillated on the other. The test specimen is gripped between the oscillating chuck (A) and the stationary chuck (B).

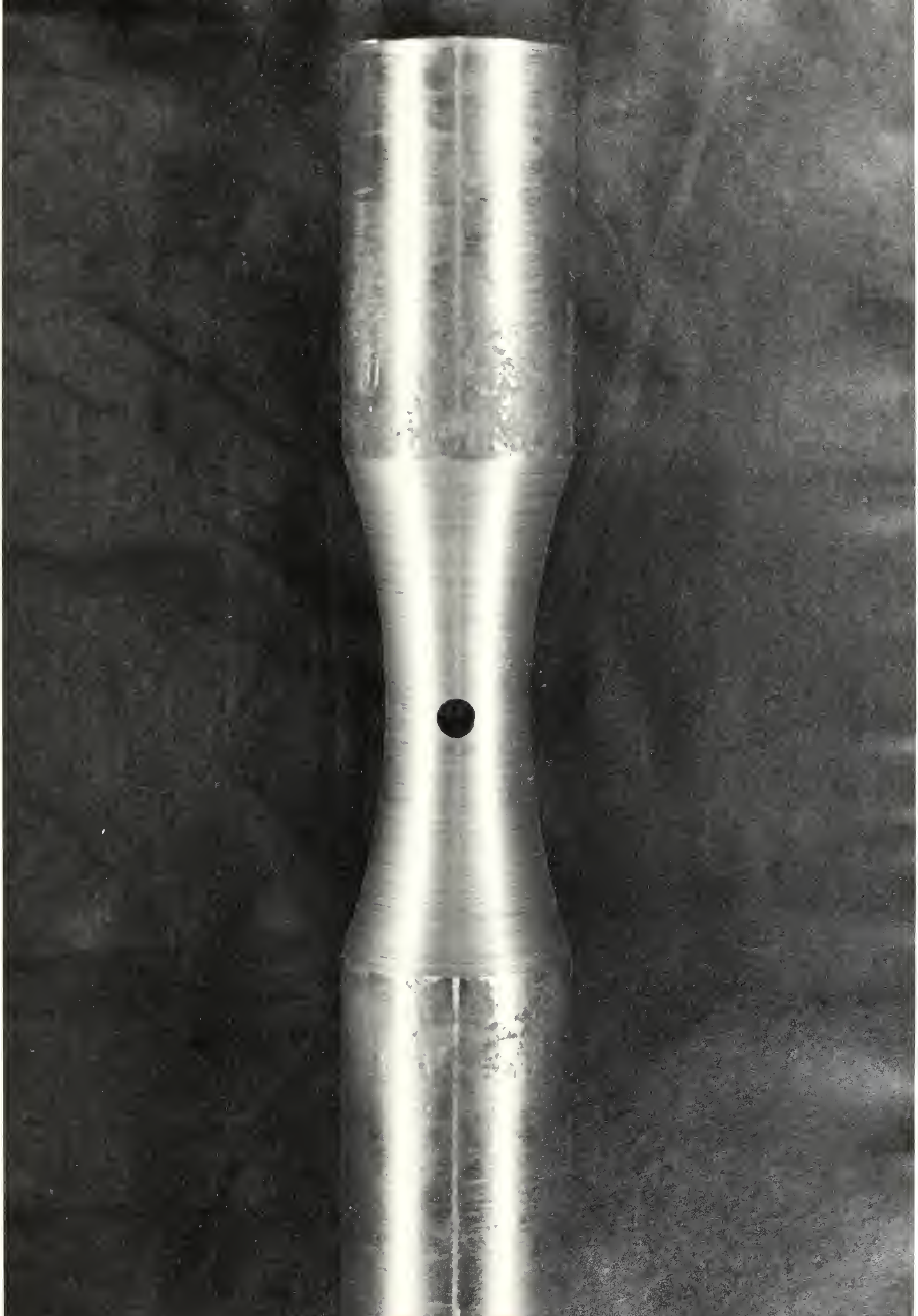


Figure 4

CUT-DOWN SPECIMEN

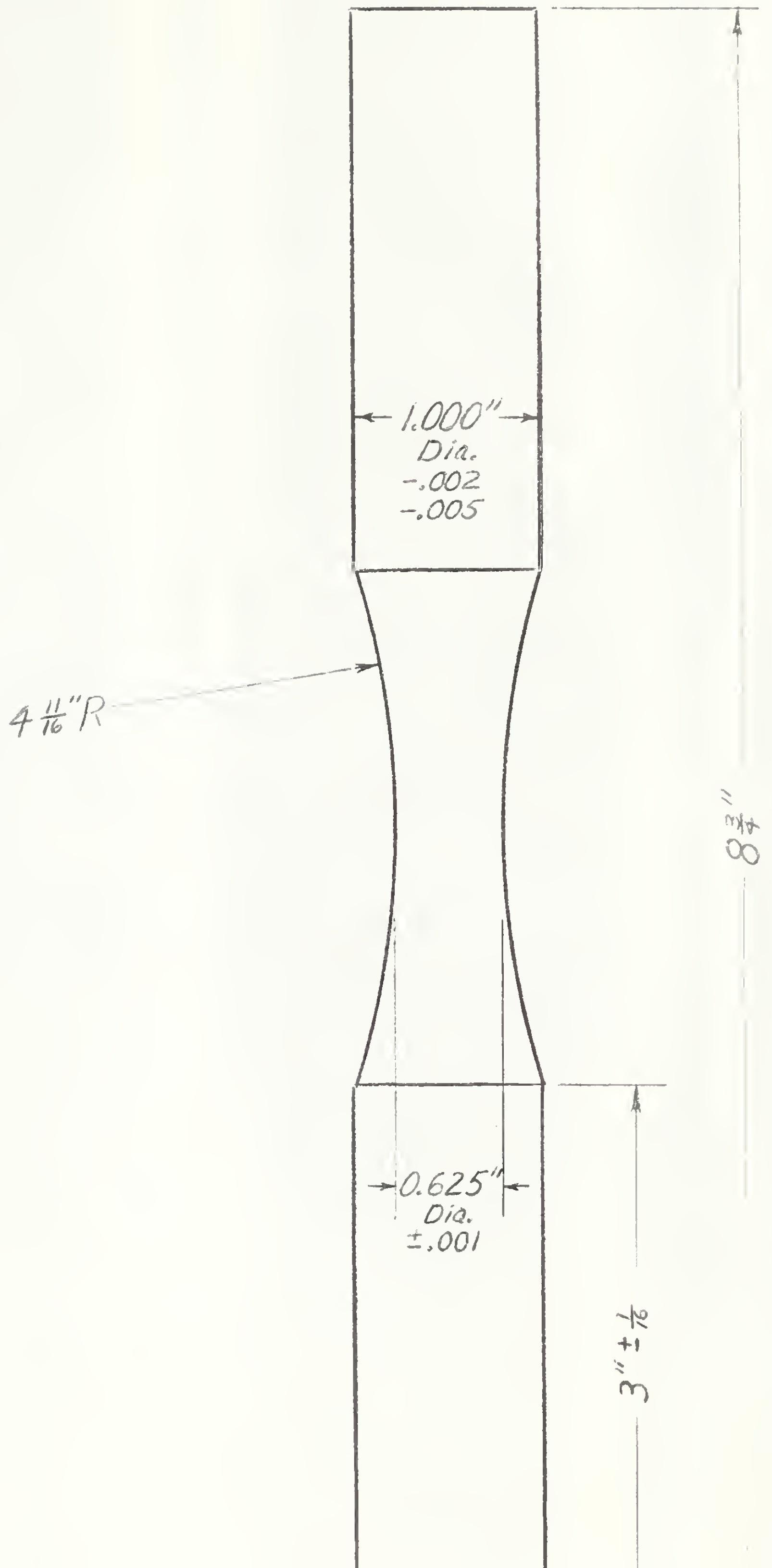


Figure 5

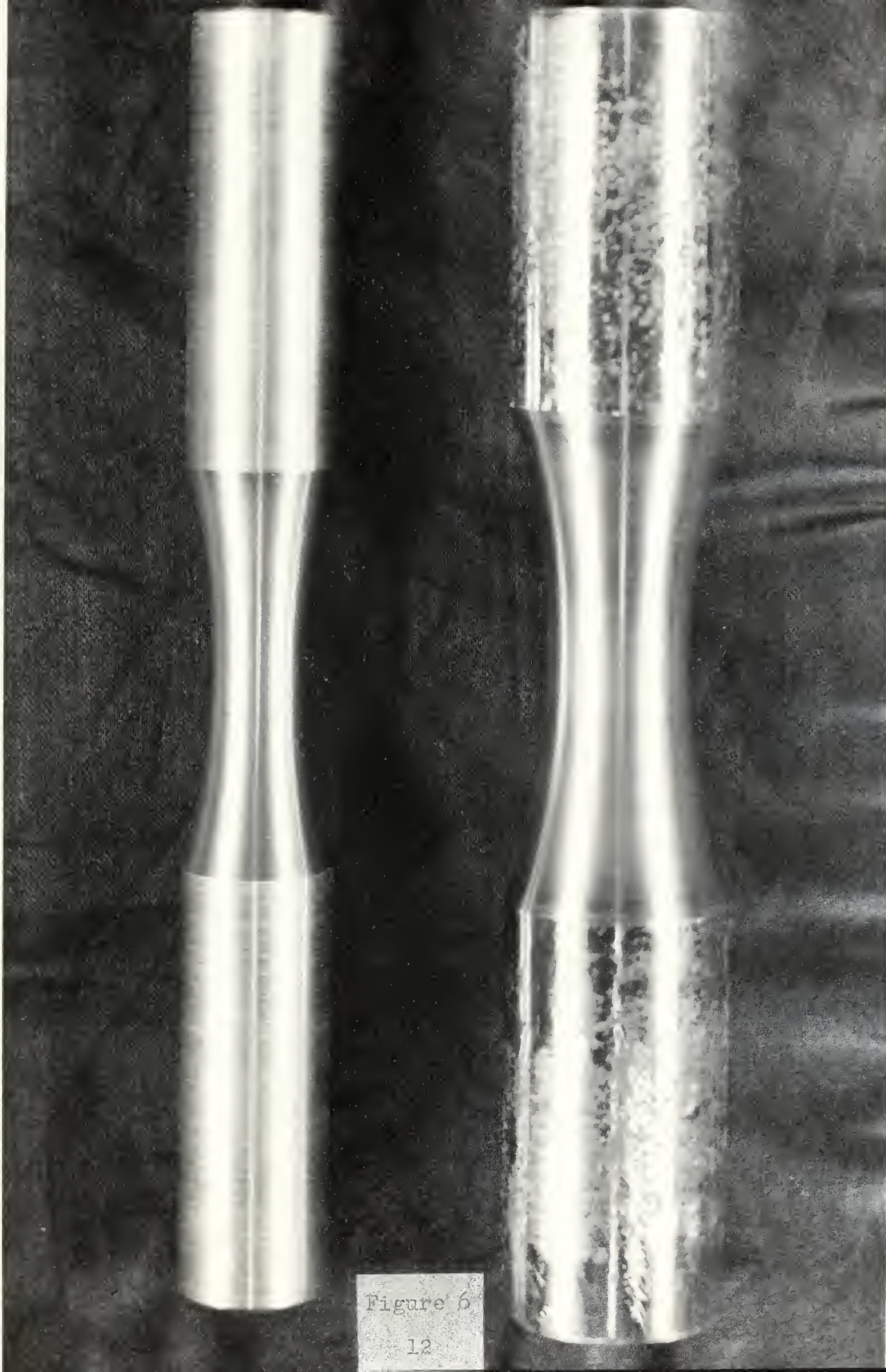


Figure 6

Figure 7

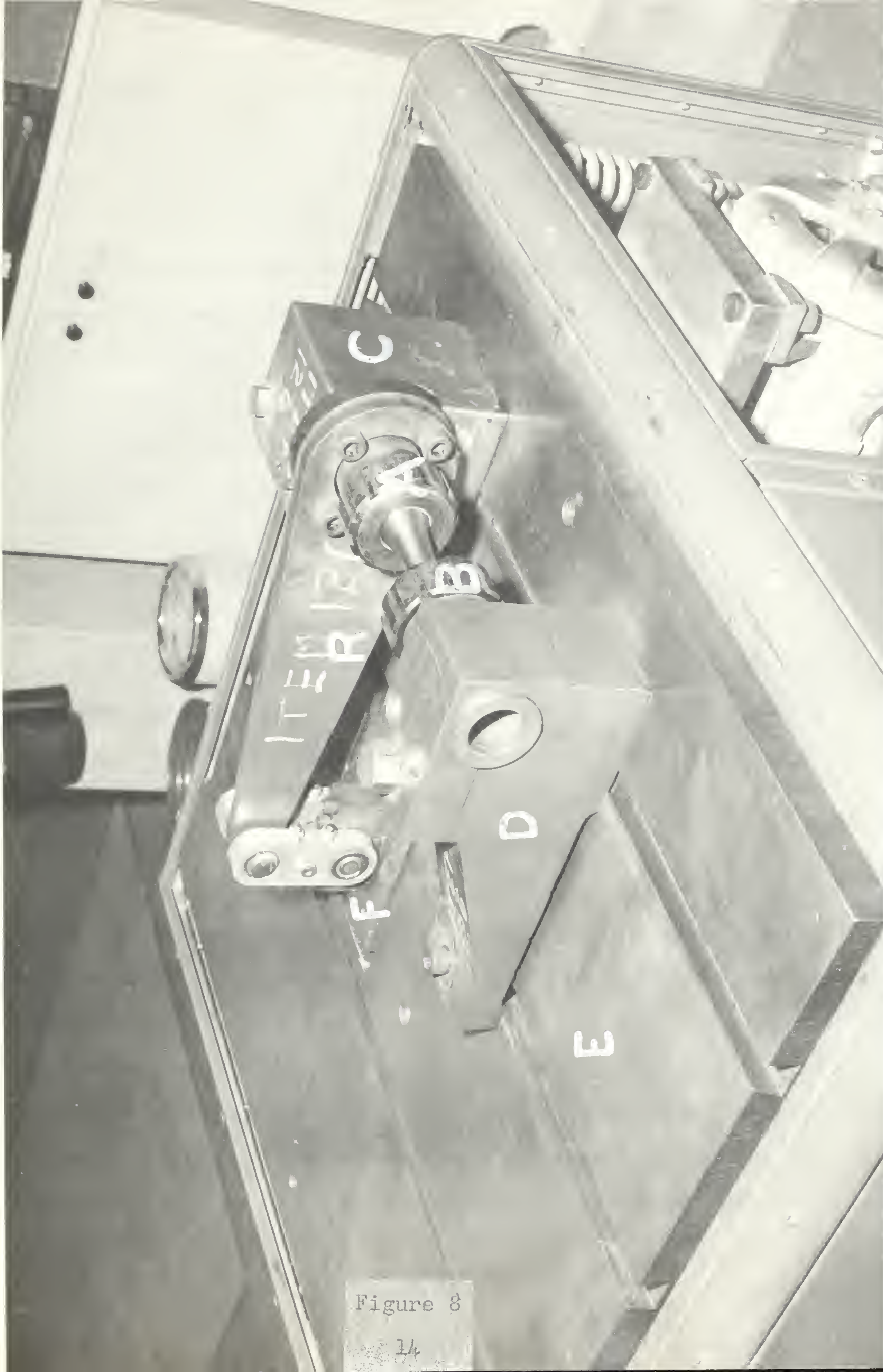


Figure 8

The oscillating chuck is supported and guided between two bearings in the pillow block (C) and oscillated by the crank. The stationary chuck is clamped rigidly on split pillow block (D). Both pillow blocks are rigidly bolted to the large stationary platen (E).

The standard torsion fixture is supplied with two crank arms. The longer, 15-1/4-inch, arm was used for all type "M" specimens and is the one shown in figure 8. It was necessary to shift to the shorter, 6-1/4-inch, arm shown in figure 9 for tests of the cut-down specimens.

The force P (lbs) required to subject a specimen of diameter D (inches) to a torsional stress τ (psi) is calculated from the equation

$$P = \frac{2J\tau}{DR} \text{ lbs.} \qquad (I)$$

where J is the polar moment of inertia of specimen area in inches^4 and R is the crank arm length in inches.

With any specimen of a given diameter, subjected to a certain repeated stress, the amplitude of vibration of the reciprocating platen depends on the specimen length. The maximum recommended amplitude of vibration Y is 0.37 inches for the type of tests conducted for this project. This amplitude can be predicted from the following equation:

$$Y = \frac{2RL\tau}{DG} \text{ in.} \qquad (II)$$

where L is the specimen free length in inches, D is the specimen diameter in inches, G is the modulus of rigidity (shear modulus) in psi, and R and τ are as in equation (I).

The machine is equipped with micro-switches which were adjusted to cut off the motor when the maximum recommended amplitude of vibration was exceeded. A counter on the machine records the number of cycles to the nearest kilocycle and automatically cuts out when the micro-switches are tripped.

Minor difficulties encountered in the use of this machine are discussed in Appendix B.

Figure 9

3. Procedure

The test procedure was chosen with a view toward obtaining the necessary data within the time and funds alloted to the project. The ASTM "Standard" Test, Method A-2, contained in STP 91-A [2] was adopted. In method A-2, each group at a particular stress level should consist of at least four specimens in order to estimate the variability of the data. Three or more different stress levels must be investigated for the determination of the curves for any particular percentage survival. Such curves (figure 13) are sometimes referred to as probability-stress-cycle (P-S-N) curves. Generally, at least four or five stress levels are used in a test of this nature. The minimum number of specimens to obtain an S-N curve was determined to be four per group at four stress levels for a total of 16 specimens. Since an S-N curve was desired for each of four configurations the total minimum requirement became 64 specimens.

Of the total of 84 specimens obtained from the total stock of four bars, two were discarded due to errors in machining, five were used in the preliminary tests, and five were discarded because fretting in the collets caused heating and/or other conditions of variability not considered satisfactory for purposes of comparison. Seven of the specimens that did not fail on original tests were cut down to smaller dimensions and tested again.

The statistical data from tests conducted on 79 specimens are contained in Appendix A.

Hardness readings were taken on the milled flats at least one diameter length from the end for uniformity. The hardness of the narrow portion of the test section was obtained on a few standard specimens and on all of the cut down specimens after failure. A standard Rockwell

hardness testing machine was used. The hardness recorded represents the average of five readings to the nearest tenth of a point.

The nominal shear stress (τ) varied between 30,000 and 65,000 psi. It was calculated from the conventional equation

$$\tau = \frac{T}{Z'} = \frac{PR}{\frac{\pi D^3}{16}} = \frac{16 PR}{\pi D^3} \text{ psi} \qquad (III)$$

for the solid specimen, where P and R are as defined in equation I, D is the minimum diameter in the test section in inches, T is the torque in inch-pounds, and Z' is the section modulus based on the polar moment of inertia in inches cubed. In order to conform to Peterson's stress concentration factors, figure 10, the stress in the specimens with the transverse hole was computed from

$$\tau = \frac{T}{Z'} = \frac{PR}{\frac{\pi D^3}{16} - \frac{dD^2}{6}} \text{ psi} \qquad (IV),$$

where d is the diameter of the transverse hole. Equation (IV) is approximate, but it is considered sufficiently accurate for these tests.

In view of the excessive scatter that results from too many variables involved in the tests [3], care was taken to maintain uniformity. As indicated previously, all specimens were taken from four bars that were certified to be from the same heat. Precautions were taken in maintaining uniform times, temperatures, and techniques in heat treatment. The machining, polishing, and drilling were carried out in such a manner that the specimens of each particular type were as nearly identical as the skill of the machinist would permit. Precautions were taken in testing the specimens to insure that there were no undesired stresses applied, such as preload, tension, or bending, which would create non-uniform testing conditions.

In spite of precautions, certain irregularities were evident such as

the variation in hardness of the heat treated specimens, slight variations in surface finish, and differences in minimum diameter between specimens. One of the causes of these variations is the improvement in technique by the machinist and the heat-treater between the first and the last specimens. This source of variation could be minimized if a machinist were fabricating these types of specimens in sufficient quantity to acquire a stable level of skill. The same holds true of the man doing the heat treating, or better still an automatic heat treating device would give readily reproducible strength and hardness.

4. Results and Discussion

In the presentation of experimental data, it is considered desirable to include where possible theoretical values or empirical data for the purpose of comparison. Figure 10 [1] shows the theoretical stress concentration factor for various values of d/D for a shaft with a transverse hole. This gives a theoretical stress concentration factor of approximately 1.73 for the specimens with 1/16-inch holes. The notch sensitivity factor of the specimen with a 1/16-inch hole is about 0.97 for quenched and tempered steel as based on Peterson's curves [1] Because the factor is nearly one, a factor of unity is used. Experimental curves giving stress concentration factors due to the rough surface finish are shown in figure 11 [4]. With the specimens indicating a Brinell hardness of about 350, a machined finish has a stress concentration factor of about 1.33. Although the smooth specimens are not mirror polished, they are so nearly so that no stress concentration factor is considered necessary.

The above predicted values of stress concentration factor compare favorably with those obtained from the 50% survival S-N curves of figure 12. The comparison is made at 10^6 cycles which is approximately the endurance limit of this material. The comparison between smooth and rough specimens gives a stress concentration factor for the machined finish of 1.31, compared with the predicted value of 1.33. The stress concentration factor for the smooth specimen with a transverse hole appears to be about 1.83 from a comparison of S-N curves at 10^6 cycles. This is reasonably close to the 1.73 value predicted. Table A presents the comparison between predicted and actual stress concentration factors for the various configurations in tabular form.

TABLE A

Specimen Type (a)	Stress Concentration Factor	
	Predicted (b)	Actual
Rough finish (2)	1.33	1.31
1/16-inch hole, smooth (3)	1.73	1.83
1/16-inch hole, rough (4)	1.33 x 1.73 = 2.30	1.83
1/4-inch hole, rough	1.33 x 1.44 = 1.92	1.74 (c)

(a) Numbers refer to the basic configurations described in section 2, and illustrated in figure 3.

(b) For the combinations of stress raisers the factors were multiplied together to obtain a conservative and recommended [4] stress concentration factor.

(c) This figure was calculated from the S-N curve of the smooth specimen and the one 50% survival point plotted for the rough specimens with a 1/4-inch transverse hole. The 50% survival data were compared at the 3 x 10^5 cycle life rather than at 10^6 cycles. As may be observed in figure 12 and other typical log log presentations of S-N curves [4] , where stress raisers are present the slope of the S-N curve is generally steeper than that of a standard smooth specimen. Therefore, the actual stress concentration factor at 10^6 cycles may be expected to be somewhat greater than the 1.74 recorded; perhaps it may compare closely with 1.92.

The curves plotted in figure 12 represent the number of stress cycles that 50% of the population would survive at a particular stress level. They are fitted to the medians of the groups at the several applied stress levels. The median, an "order statistic", is the middlemost value when the observed values are arranged in order of magnitude, or the average of the two middlemost values if the group size is even.

This procedure for analysis of data was adopted from ASTM STP 91-A [2]. The following statement is quoted in reference to the above procedure:

These techniques should be used when the actual shape of the distribution of fatigue life values for a material is unknown, or sketchy, and the number of specimens tested at each applied stress level is too small, say less than 50, to estimate the shape of the distribution. In such cases these techniques give conservative results.

Figure 13 shows the probability-stress-cycle curves for one configuration. These percentages represent median percentages of survivors for the population and are based on the number of specimens tested at each stress level. They are called "median percentages" because 50% of the time the true percentage will be larger and 50% of the time it will be smaller; i.e., the confidence level is 50%. Considering the limited number of specimens tested at each stress level and considering the objectives of this project, we conclude that the information to be obtained from this graphical presentation is limited. Therefore, no attempt has been made to compare the fatigue life of the various configurations by use of other than the 50% survival curves.

SHEAR STRESS CONCENTRATION FACTOR, K_t, FOR THE TORSION CASE OF A SHAFT WITH A TRANSVERSE HOLE

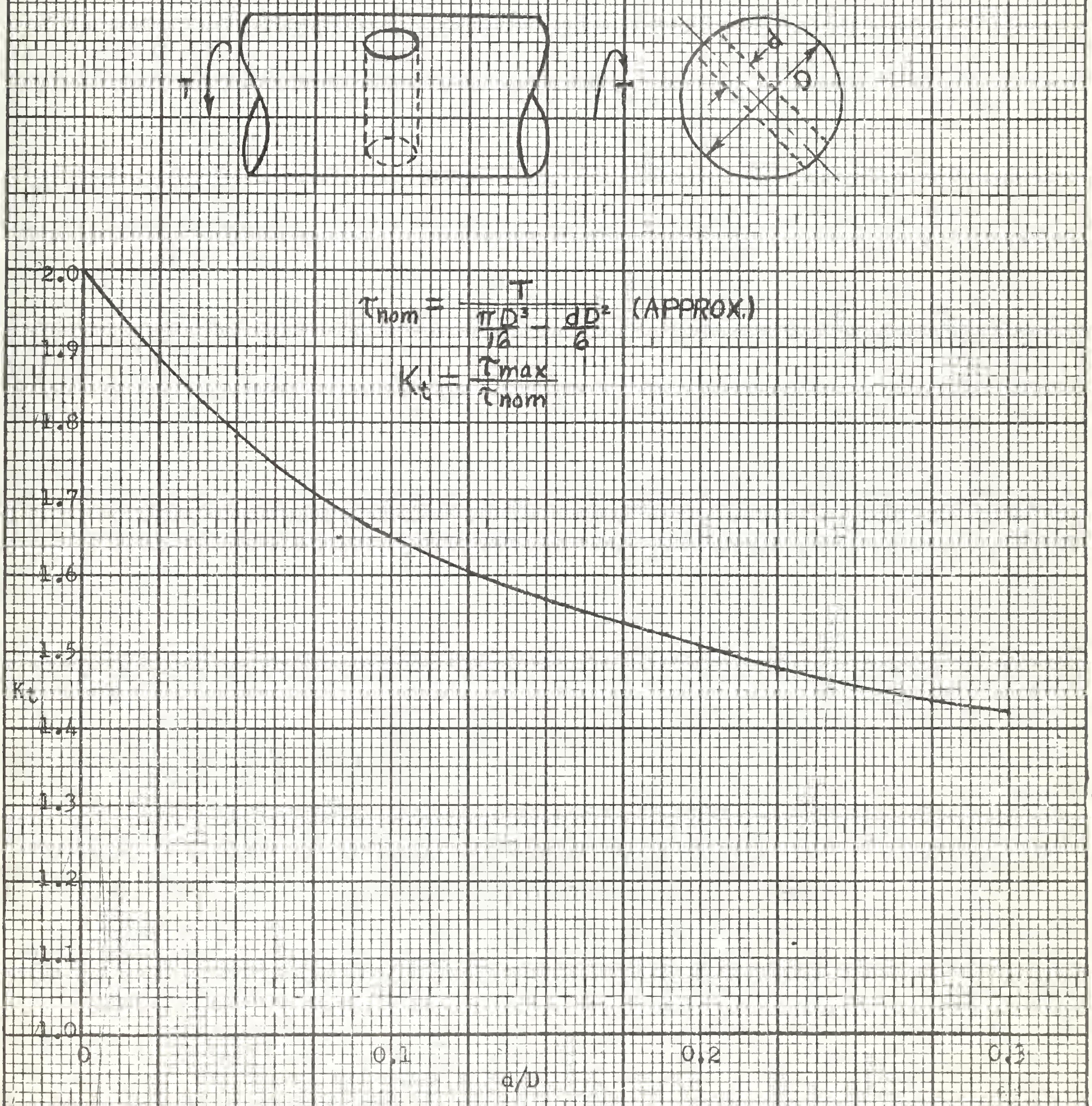


Figure 10

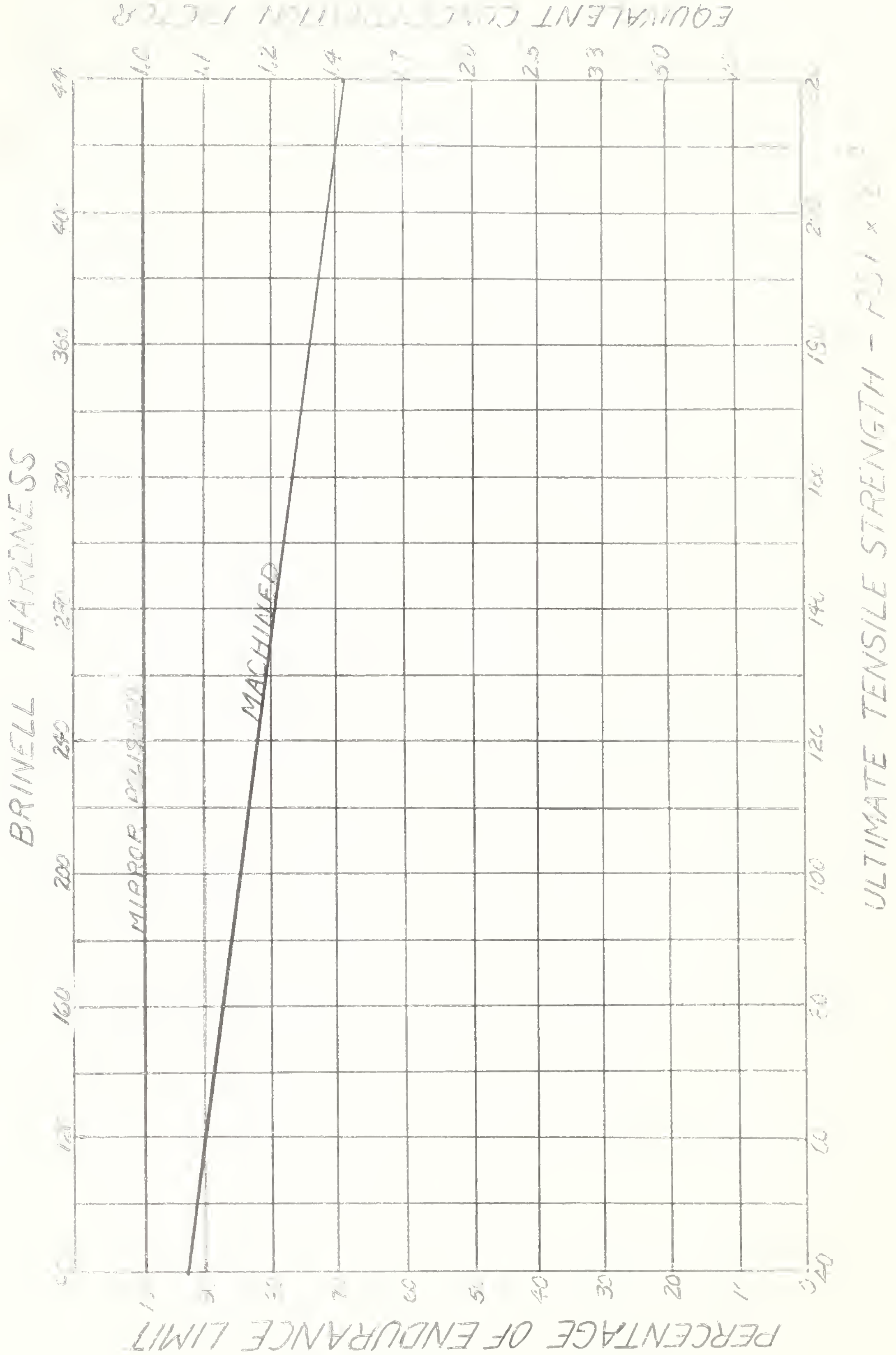

BRINELL HARDNESS
ULTIMATE TENSILE STRENGTH
EQUIVALENT
PERCENTAGE OF ENDURANCE LIMIT
MACHINED
MIRROR

Figure 11

S-N CURVES FOR 50% SURVIVAL

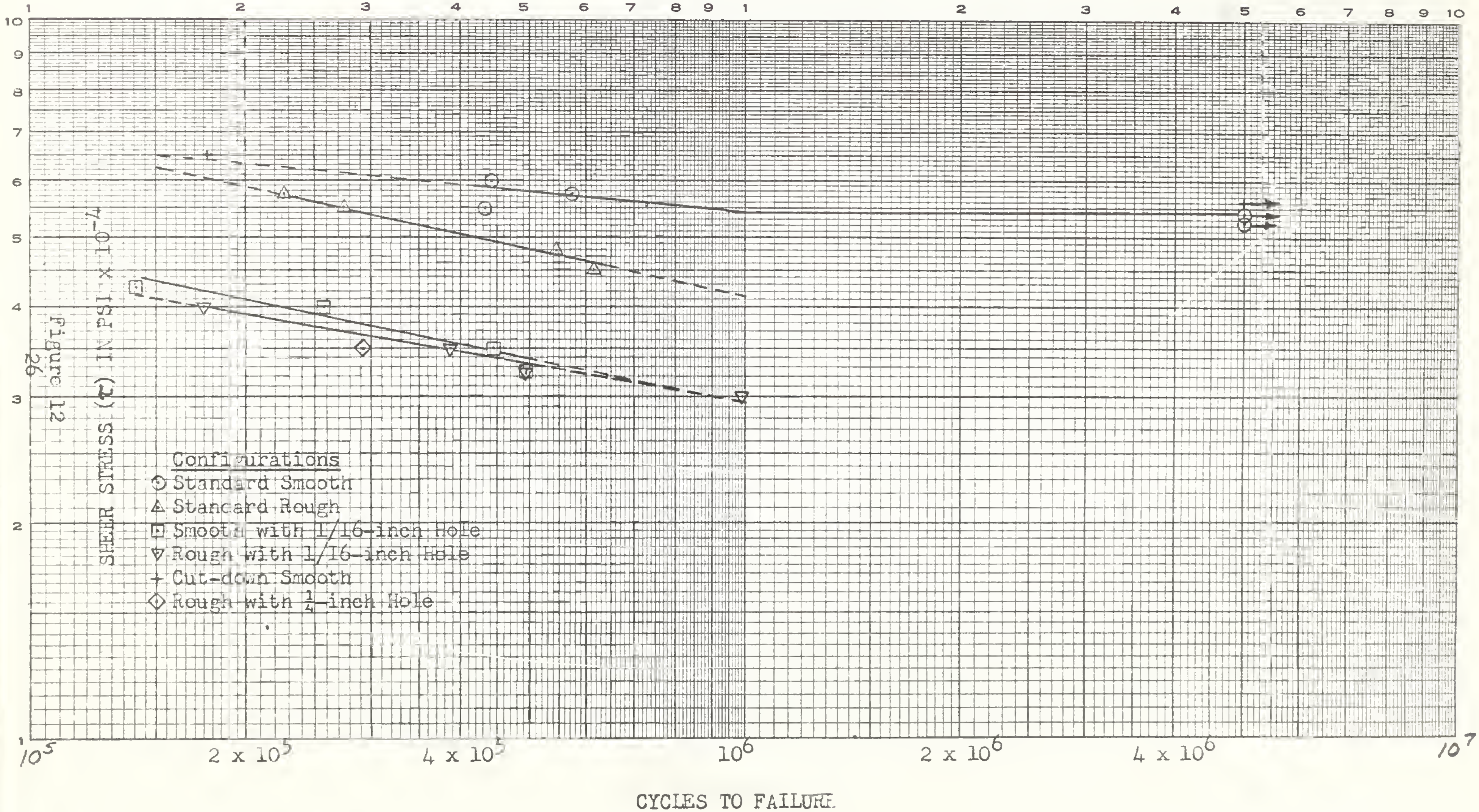


Figure 12

PROBABILITY-STRESS-CYCLE CURVE
FOR A ROUGH SPECIMEN WITH 1/16-INCH HOLE

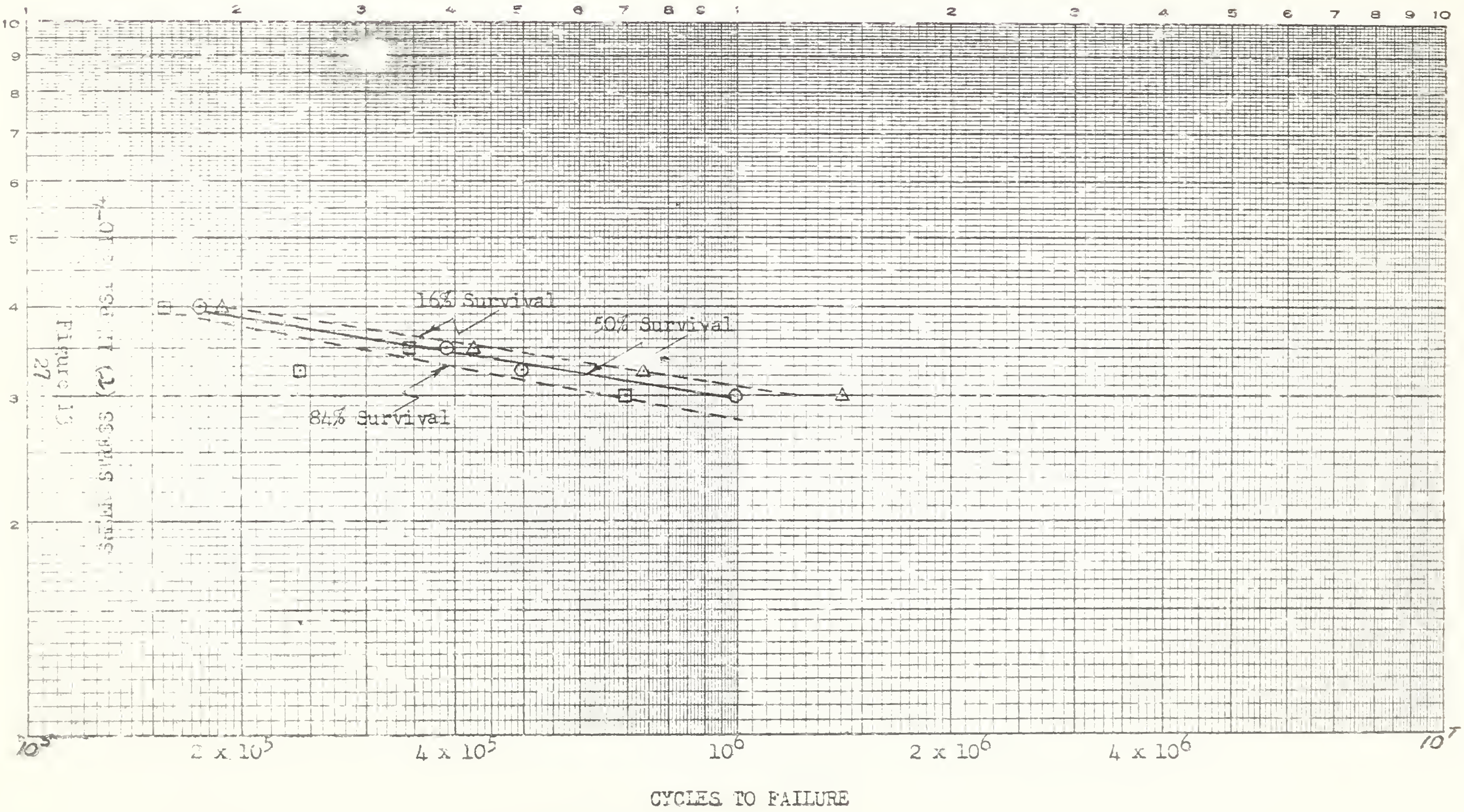


Figure 13

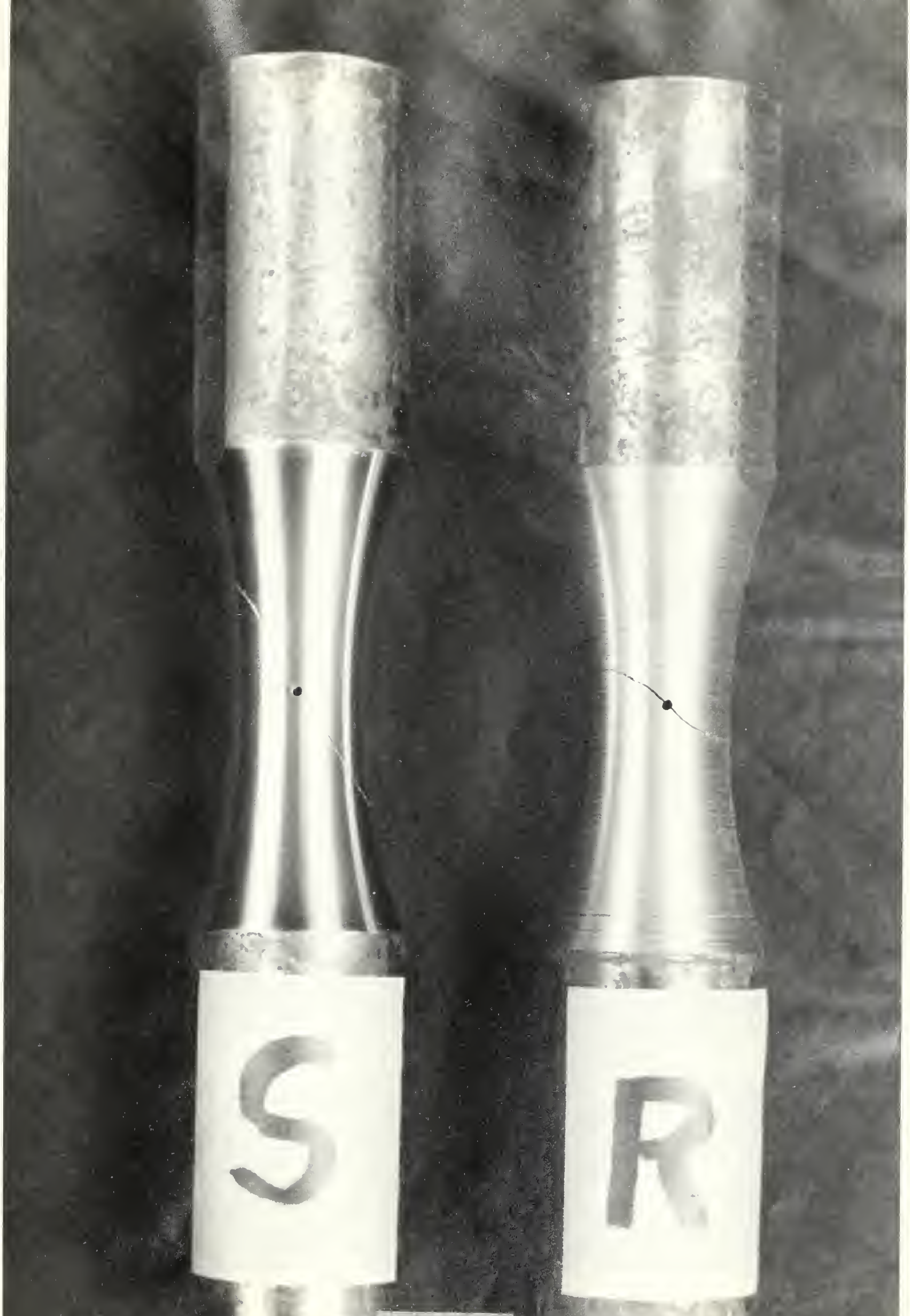


Figure 14

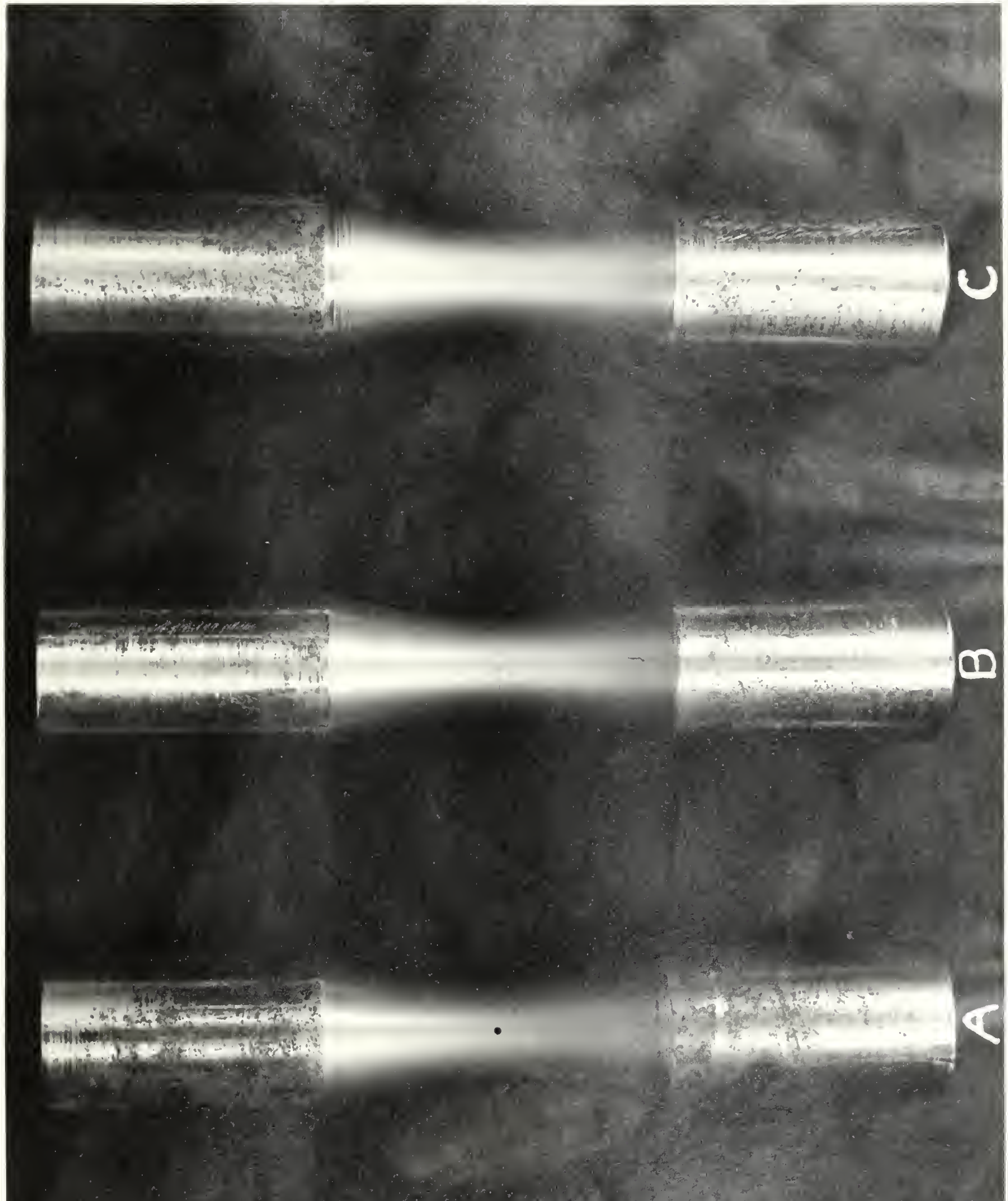


Figure 12

One of the notable variables of the specimens is the hardness. To further examine this area, we made a bar chart showing the frequencies of hardness within a range of 0.5 points on the Rockwell "C" scale, figure 16. The bar chart naturally does not quite follow the normal bell shaped distribution curve, but is about as close as could be expected with the limited number of specimens (a total of 72). There is only one group for which it is obvious that hardness may have had an effect on the 50% survival point, namely the standard smooth specimens tested at 55,000 psi, Appendix A, Table I. Specimens A-12 and A-13 were three and four Rockwell "C" points softer than the other two and they failed significantly earlier than the harder two. This in itself would not be significant considering the wide scatter that can be expected when testing in the vicinity of the endurance limit of a material; but the 50% survival point obtained from the group tested at 55,000 psi falls significantly below the 50% survival curve for the standard smooth configuration.

There are two points along the S-N curve for the standard smooth specimens, figure 12, that were obtained from the seven cut-down specimens mentioned on page 9. Some factors that may have effected the strength of these smaller specimens are: the Rockwell "C" hardness is about two points lower than the larger specimens; the size effect gives a slight strength advantage to the smaller specimens [4] ; understressing for 5 million cycles may have had a strengthening effect. These effects apparently cancelled each other because the points mentioned above agree closely with the curve plotted from standard size specimens.

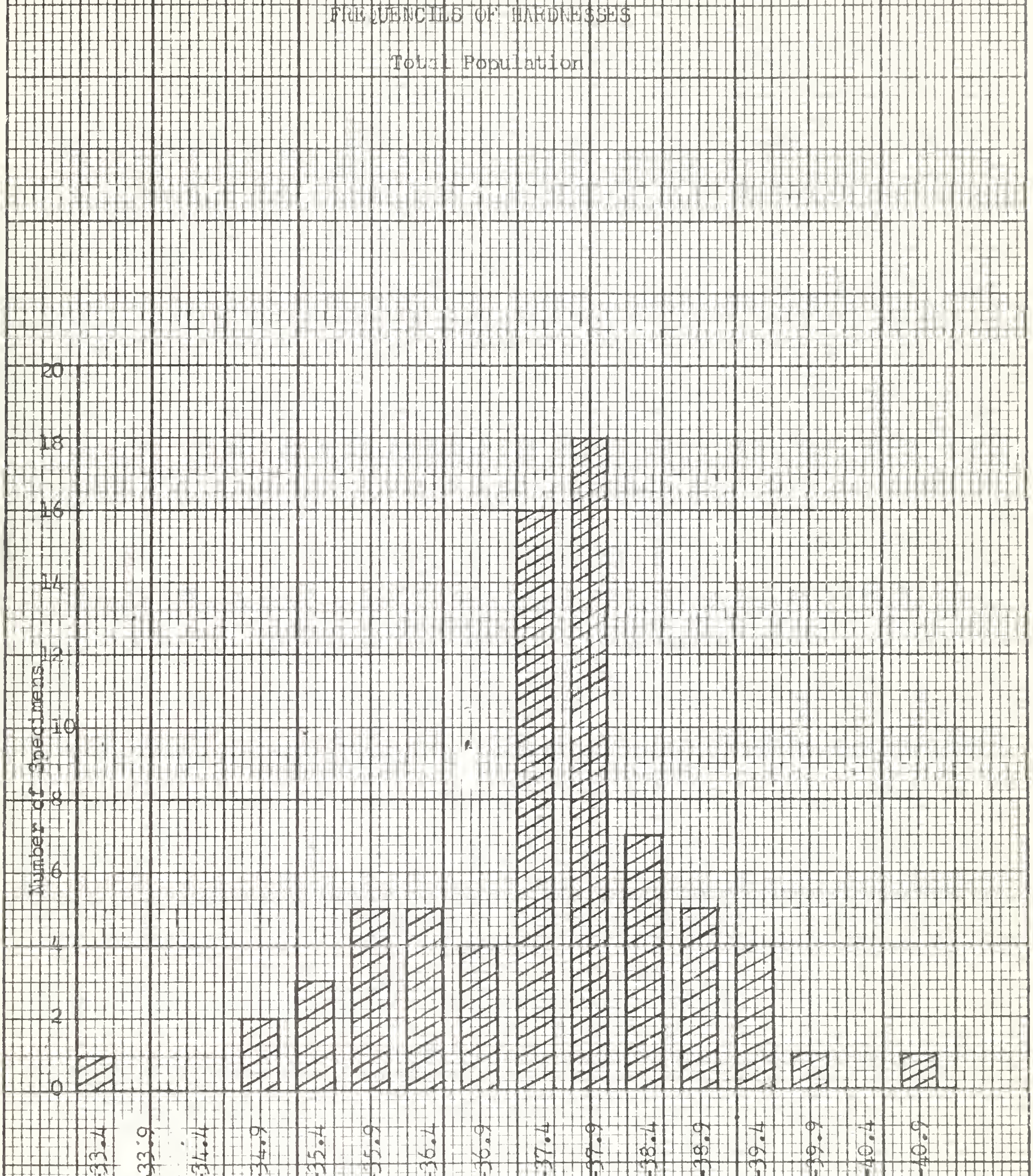

FREQUENCIES OF HARDNESSES
Total Population
Number of Specimens
20
18
16
14
12
10
8
6
4
2
0
33.0-33.4
33.5-33.9
34.0-34.4
34.5-34.9
35.0-35.4
35.5-35.9
36.0-36.4
36.5-36.9
37.0-37.4
37.5-37.9
38.0-38.4
38.5-38.9
39.0-39.4
39.5-39.9
40.0-40.4
40.5-40.9
Rockwell "C" Hardness

Figure 15

5. Conclusions

In evaluating results obtained from fatigue tests, one recognizes that the analysis is of a statistical nature. Furthermore, the application of statistical methods to the analysis of the test results of certain configurations offers a means only for estimating the characteristics of the aggregate of these configurations [2] .

In light of the above statements, the following conclusions are drawn from the results obtained in this project (see Table A, page 22):

a. The stress concentration factor for a round specimen with a rough surface finish, type 2, subjected to reversed stresses in torsion, is verified to be approximately as predicted from figure 11.

b. The stress concentration factor for a round specimen with a 1/16-inch hole, type 3, subjected to reversed stresses in torsion, is verified to be approximately as predicted from figure 10.

c. The actual stress concentration factor, for a 1/16-inch hole and a rough surface finish combined, is not as great as would be predicted by multiplying the two individual stress concentration factors together. In the case in which the stress concentration factor for the hole was considerably greater than that for the rough finish, it is possible that the larger stress raiser predominates. The major effect of the rough surface finish in this combination appears to be to hasten the failure once a crack has started. This effect may be observed in figure 17 by comparing curve 2, which was taken from a smooth specimen, with either of the other three, which were obtained from rough specimens.

d. The actual stress concentration factor for a 1/4-inch hole and a rough surface finish (as indicated from a very small number

of tests) is somewhat less than the value predicted by multiplying the two individual stress concentration factors together, but it appears that the total effect is more nearly equal to the product of the factors than is true for the smaller hole and rough surface.

e. If the indications in paragraph c above are correct, it seems that where two stress raisers appear at a point and one stress concentration factor is considerably larger, the larger one tends to dominate, and the effect of the smaller stress raiser will be less than its stress concentration factor would indicate. Perhaps applying a stress concentration factor obtained from that of the larger stress raiser increased by 10% would give a good prediction for this combination. Also, if the indications in paragraph d are correct, it seems appropriate for conservative design to multiply the stress concentration factors together to obtain a prediction of the effect of two stress raisers of similar value acting together at a point.

The tool marks or scratches which constituted the rough machine finish on specimens used in this project were all circular. No attempt was made to determine the effect of a similar finish consisting of longitudinal scratches. However, there is information in the literature [5], [6] comparing the effects of circular versus longitudinal scratches under various conditions of stress.

It is recommended that further studies be conducted on this problem using a wide variety of stress raiser combinations in bending and push-pull loads as well as torsion, to ascertain whether the indications of these tests are valid, whether they are applicable to stress raiser

combinations in general, and whether the proposed methods of predicting stress concentration factors are applicable to bending and push-pull loads as well as torsion.

It is further recommended that if future tests are to be made on the Baldwin Universal Fatigue Testing Machine, the problem of failure criterion discussed in Appendix B be considered. Proper consideration in specimen design can provide sufficient allowance for amplitude change before cut-off to prevent the artificial failure conditions experienced in this project.

6. Bibliography

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STANDARD SMOOTH SPECIMENS

Specimen Number	Minimum Diameter (Inches)	Rockwell "C" Hardness	Load Applied $lbs/in.^2$	Cycles to Failure	Amplitude Change to Failure	Cycles to Failure for 50% Probability	Remarks
A-4	.939	36.4	52,500	2,100,000 No Failure	-		Test Stopped Before Failure
B-2	.936	37.6	52,500	5,000,000 No Failure			" "
B-3	.939	37.4	52,500	5,000,000 No Failure		5,000,000 +	" "
B-4	.938	37.9	52,500	5,000,000 No Failure			" "
B-6	.940	38.3	54,000	5,000,000 No Failure			" "
B-8	.940	38.1	54,000	5,000,000 No Failure		5,000,000 +	" "
B-10	.940	37.1	54,000	5,000,000 No Failure			" "

Table 1

STANDARD SMOOTH SPECIMENS

Specimen Number	Minimum Diameter (Inches)	Rockwell "C" Hardness	Load Applied lbs/in^2	Cycles to Failure	Amplitude Change to Failure	Cycles to Failure for 50% Probability	Remarks
A-13	.936	33.4	55,000	191,000	0.05"		Heated to Bluish color in test area.
A-18	.940	37.5	55,000	3,995,000	0.05"	433,500	
A-12	.938	34.8	55,000	269,000	0.05"		
A-19	.938	37.5	55,000	598,000	0.05"		
B 18	.940	37.1	57,500	790,000	0.03"		
B-21	.939	37.7	57,500	5,000,000 No failure		574,000	Test Stopped before Failure
C-13	.938	38.2	57,500	358,000	0.03"		
C-15	.938	38.9	57,500	146,000	0.03"		

Table 1

STANDARD SMOOTH SPECIMENS

Specimen Number	Minimum Diameter (inches)	Rockwell "C" Hardness	Load Applied lbs/in^2	Cycles to Failure	Amplitude Change to Failure	Cycles to Failure for 50% Probability	Remarks
C-18	.939	38.8	60,000	189,000	0.03"		
C-21	.939	35.8	60,000	194,000	0.03"		
D-1	.938	34.8	60,000	441,000	0.03"	441,000	
D-14	.940	35.8	60,000	515,000	0.03"		
D-15	.940	35.8	60,000	1,141,000	0.03"		

Table 1

SMALL SMOOTH SPECIMENS

Specimen Number	Minimum Diameter (inches)	Rockwell "C" Hardness	Load Applied lbs/in^2	Cycles to Failure	Amplitude Change to Failure	Cycles to Failure for 50% Probability	Remarks
B-10	.626	33.8	56,000	1,453,000	0.23"		
B 21	.627	37.5	56,000	5,000,000		5,000,000+	No failure
D-10	.626	32.4	56,000	5,000,000			No failure
B-2	.626	33.6	65,000	138,000	0.21"		
B-4	.626	35.9	65,000	107,000	0.21"	177,000	
B-6	.626	34.8	65,000	216,000	0.21"		
B-8	.624	35.3	65,000	236,000	0.21"		

Table 1

ROUGH MACHINED SPECIMENS

Specimen Number	Minimum Diameter (inches)	Rockwell "C" Hardness	Load Applied lbs/in^2	Cycles to Failure	Amplitude Change to Failure	Cycles to Failure for 50% Probability	Remarks
D-10	.939	38.3	45,000	5,000,000 No Failure			Test Stopped Before Failure
D-13	.939	37.1	45,000	608,000	0.11"	616,000	
D-19	.939	38.4	45,000	624,000	0.11"		
D-20	.936	37.5	45,000	602,000	0.11"		
A-15	.940	36.3	48,000	430,000	0.10"		
A-14	.939	35.0	48,000	411,000	0.10"	546,500	
A-16	.941	39.4	48,000	663,000	0.10"		
A-21	.941	39.1	48,000	671,000	0.10"		
B-13	.941	37.2	55,000	327,000	0.07"		
B-20	.941	35.8	55,000	272,000	0.07"	273,500	
B-19	.939	37.3	55,000	275,000	0.07"		
C-16	.939	37.8	55,000	254,000	0.07"		

Table 2

ROUGH MACHINED SPECIMENS

Specimen Number	Minimum Diameter (inches)	Rockwell "C" Hardness	Load Applied lbs/in^2	Cycles to Failure	Amplitude Change to Failure	Cycles to Failure for 50% Probability	Remarks
D-4	.938	35.3	57,500	257,000	0.05"		
D-6	.939	35.8	57,500	212,000	0.05"		
D-17	.938	37.0	57,500	168,000	0.05"	226,500	
D-21	.939	35.3	57,500	241,000	0.05"		

Table 2

1/16" HOLE SMOOTH SPECIMENS

Specimen Number	Minimum Diameter (inches)	Rockwell "C" Hardness	Load Applied lbs/in^2	Cycles to Failure	Amplitude Change to Failure	Cycles to Failure for 50% Probability	Remarks
C-3	.939	40.5	32,500	592,000	0.21"		
C-12	.940	37.0	32,500	947,000	0.21"	496,000	
C-14	.939	37.9	32,500	272,000	0.21"		
C-20	.940	37.5	32,500	400,000	0.21"		
A-17	.936	38.7	35,000	593,000	0.19"		
A-8	.941	38.0	35,000	467,000	0.19"	445,000	
A-9	.939	36.2	35,000	385,000	0.19"		
A-10	.939	39.1	35,000	423,000	0.19"		

Table 3

1/16" HOLE SMOOTH SPECIMENS

Specimen Number	Minimum Diameter (inches)	Rockwell "C" Hardness	Load Applied lbs/in^2	Cycles to Failure	Amplitude Change to Failure	Cycles to Failure for 50% Probability	Remarks
B-5	.939	37.8	40,000	238,000	0.16"		
B-7	.939	37.9	40,000	277,000	0.16"	257,500	
B-17	.942	37.9	40,000	285,000	0.16"		
D-12	.944	37.0	40,000	217,000	0.16"		
D-7	.938	37.9	42,500	140,000	0.15"		
D-8	.937	38.3	42,500	97,000	0.15"	141,000	
D-16	.940	36.9	42,500	162,000	0.15"		
D-18	.940	37.2	42,500	142,000	0.15"		

Table 3

1/16" HOLE ROUGH SPECIMENS

Specimen Number	Minimum Diameter (inches)	Rockwell "C" Hardness	Load Applied lbs/in^2	Cycles to Failure	Amplitude Change to Failure	Cycles to Failure fo. 50% Probability	Remarks
A-7	.938	39.9	30,000	1,225,000	0.22"		
A-11	.940	39.2	30,000	697,000	0.22"		
A-20	.940	38.9	30,000	758,000	0.22"	991,500	
B-11	.938	37.0	30,000	1,408,000	0.22"		
C-9	.940	38.6	32,500	633,000	0.21"		
C-10	.939	37.8	32,500	739,000	0.21"	497,500	
C-19	.939	37.7	32,500	242,000	0.21"		
D-9	.939	36.9	32,500	362,000	0.21"		

Table 4

1/16" HOLE ROUGH SPECIMENS

Specimen Number	Minimum Diameter (inches)	Rockwell "C" Hardness	Load Applied lbs/in^2	Cycles to Failure	Amplitude Change to Failure	Cycles to Failure for 50% Probability	Remarks
B-16	.939	37.8	35,000	415,000	0.19"		
B-12	.944	36.7	35,000	425,000	0.19"	389,000	
B-15	.944	36.0	35,000	346,000	0.19"		
B-9	.944	37.7	35,000	363,000	0.19"		
C-2	.940	37.0	40,000	168,000	0.17"		
C-5	.940	37.2	40,000	156,000	0.17"	175,000	
C-6	.939	37.3	40,000	188,000	0.17"		
C-8	.940	37.9	40,000	182,000	0.17"		

Table 4

1/4" HOLE ROUGH SPECIMENS

Specimen Number	Minimum Diameter (inches)	Rockwell "C" Hardness	Load Applied lbs/in^2	Cycles to Failure	Amplitude Change to Failure	Cycles to Failure for 50% Probability	Remarks
D-2	.937	37.4	35,000	314,000	0.22"		
D-3	.938	36.5	35,000	244,000	0.22"	291,500	
D-5	.937	37.3	35,000	287,000	0.22"		
D-11	.940	36.0	35,000	296,000	0.22"		

Table 5

APPENDIX B

Machine Difficulties

A few minor difficulties were experienced in testing the large torsion specimens. The high torque required to cause the solid specimens to fail taxed the ability of the collets to grip the specimens without allowing slippage. Difficulty was experienced early in the tests with specimens fretting in the collets. This fretting resulted in fusion of metal from the specimen onto the collets causing their inner surfaces to become rough. The rough surfaces in turn permitted only small areas of contact with the specimen, thereby increasing the fretting and fusion and eventually rendering the collets useless. They were repaired by a honing operation, rotating the collects on a 1-1/2-inch brass rod to which grinding compound had been applied. Once the collets were restored, further fretting was kept to a minimum by applying greater torque during tightening.

During the latter portion of the testing period, the machine was at times considerably noisier than normal. Although the increased noise pointed to the possibility of a faulty bearing in the torsion fixture, since it was intermittent and no variation in vibration amplitude was evident, tests were completed without dismantling the fixture. There are no indications that the noisy operation affected the fatigue life of the specimens tested under those conditions.

The failure criterion used with this machine created a discrepancy in the number of cycles to failure between the tests at maximum and minimum loads. In the case of maximum load tests, the limit switches cut off the motor and the specimen was considered to have failed after an

increase in amplitude of vibration of perhaps 0.02 inches. However, with the minimum load tests, the initial amplitude of vibrations was much lower and allowed an increase in amplitude of perhaps 0.22 inches before the machine was stopped by the limit switches. In an attempt to analyze this problem, several specimens were observed and data recorded during the final stage before failure. The curves of amplitude versus increase in number of cycles (figure 17) are of an exponential nature, thus minimizing the adverse effect of this artificial criterion of failure. From observation of many specimens, it was noted that there is no appreciable amplitude change until the final stages of fatigue failure.

No correction has been applied to the test data of this project, since any factor applied would be of questionable reliability. The number of cycles from a change of amplitude of 0.02 inches, which was the minimum increase in amplitude for failure at high loads, to failure of lightly loaded specimens varied from 41,000 cycles for curve 2 of figure 17 to 10,500 cycles for curve 1.

APPENDIX C

Sample Calculations

Equations III and IV, page 19, permit calculation of the nominal shear stress (τ) when other factors are known. Calculations were made to determine the force (P) to be applied to obtain a desired nominal shear stress. For these calculations equations III and IV were rearranged as follows:

Specimen A-4 (Equation III)

$$\tau = \frac{16PR}{\pi D^3}$$

$$P = \frac{\pi D^3 \tau}{16R}$$

$$= \frac{(3.14)(.939\,in)^3(52{,}500\ \frac{lbs}{in^2})}{(16)(15.25\,in)}$$

$$= 560\ lbs$$

Specimen A-17 (Equation IV)

$$\tau = \frac{PR}{\frac{\pi D^3}{16} - \frac{dD^2}{6}}$$

$$P = \frac{\tau}{R}\left(\frac{\pi D^3}{16} - \frac{dD^2}{6}\right)$$

$$= \frac{35{,}000\ \frac{lbs}{in^2}}{15.25\,in}\left[\frac{(3.14)(.936\,in)^3}{16} - \frac{(.0625\,in)(.936\,in)^2}{6}\right]$$

$$= 349\ lbs$$

Determination of the 50% survival value from each group of specimens tested at the same stress level is done by taking the middle most value of fatigue life when the group contains an odd number of specimens

or, as was the case for most of the groups tested in this project, taking the average of the two middlemost values if the group size is even.

EXAMPLE: Group of standard smooth specimens tested at 57,500 psi.

Specimen No.	Cycles to failure
C-15	146,000
C-13	358,000
B-18	790,000
B-21	5,000,000 (no failure)

$$\frac{358{,}000 + 790{,}000}{2} = 574{,}000 \text{ cycles}$$